LATERAL VIBRATION CREATED BY TORSIONAL COUPLING OF A CENTRIFUGAL COMPRESSOR SYSTEM DRIVEN BY A CURRENT SOURCE DRIVE FOR A VARIABLE SPEED INDUCTION MOTOR

by

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James H. Hudson, Consultant for A-C Compressor Corporation, began his career with Allis Chalmers Corporation in the Compressor Division (1965 to 1985). Mr. Hudson served in many capacities during that period, including Service and Erection Engineer, Associate Design Engineer, Application Engineer, Chief Engineer Single Stage Compressors, Supervisor Mechanical Engineering Custom Compressors, and Product Manager. In 1985, A-C Compressor Corporation purchased the Compressor Division from Allis Chalmers, and he assumed the position of Manager of Engineering. He assumed his current position in 1987.

Mr. Hudson graduated with a B.S.M.E. degree from Newark College of Engineering (1965). He has been a task force member on the 4th and 5th editions of the API 617 Specifications for Centrifugal Compressors and the API task force on Quality Improvement. Currently, he is a member of the 6th edition API 617 task force and a member of the API task force on rotordynamics. He has participated in previous symposia as a discussion leader on the topics of compressor maintenance and compressor performance. He is a registered Professional Engineer in the State of Wisconsin.

ABSTRACT

A serious radial gear vibration was determined to be caused by significant torsional excitation from a squirrel cage induction motor that was controlled by a variable frequency current source inverter drive. Field test data verified and quantified the frequency and magnitude of torque due to interharmonic distortion and ultimately provided data to verify that system modifications had eliminated the gear vibration and verified suitability of the system for long term variable speed operation.

INTRODUCTION

A sulfuric acid plant was designed and constructed to reclaim spent sulfuric acid used in a chemical process at the same site. The feed to the sulfuric acid plant was dependent upon the output of the process that used the sulfuric acid. It was anticipated that the quantity of the spent acid would vary through the year as a function of the output of the base process. In order to accommodate the variation of the spent acid, the new acid plant was designed to accommodate the varying quantity of spent acid as efficiently as possible.

A key mechanical component in the sulfuric acid plant is the main process blower. The blower is a single stage centrifugal compressor of the overhung design, producing a head level in the range of 15,000 to 20,000 ft. The weight flow of the compressor is directly proportional to the output of the sulfuric acid plant. Seeing how the feed to the sulfuric acid plant would vary considerably, the design of the compressor system was chosen which would optimize plant economics. As the weight flow of the acid plant varied, so did the head requirements of the sulfuric acid plant. The compressor was chosen to be an electric motor drive. If a fixed speed motor had been used, the only way to vary the pressure rise across the compressor would be to throttle either at the inlet or the discharge. A more efficient way to vary the head output of the compressor would have been to use inlet guide vanes to create prewhirl prior to the compressor inlet nozzle. The typical characteristics of a single stage compressor with inlet guide vanes are shown in Figure 1. Neither throttling nor use of inlet guide vanes

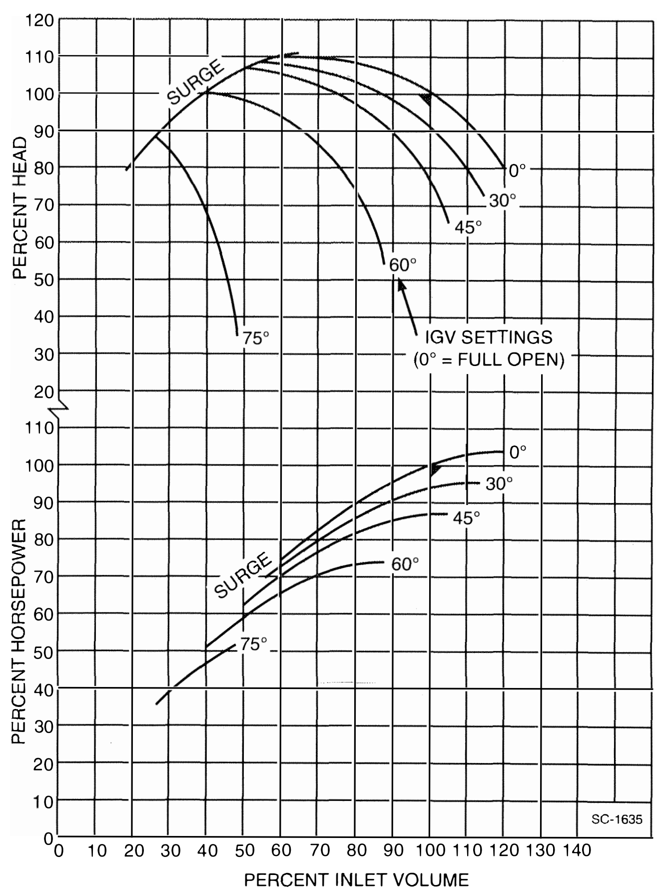


Figure 1. Typical Single Stage Compressor Performance Curve with Inlet Guide Vanes.

was as efficient as variable speed control. Compressor characteristics with variable speed control are shown in Figure 2.

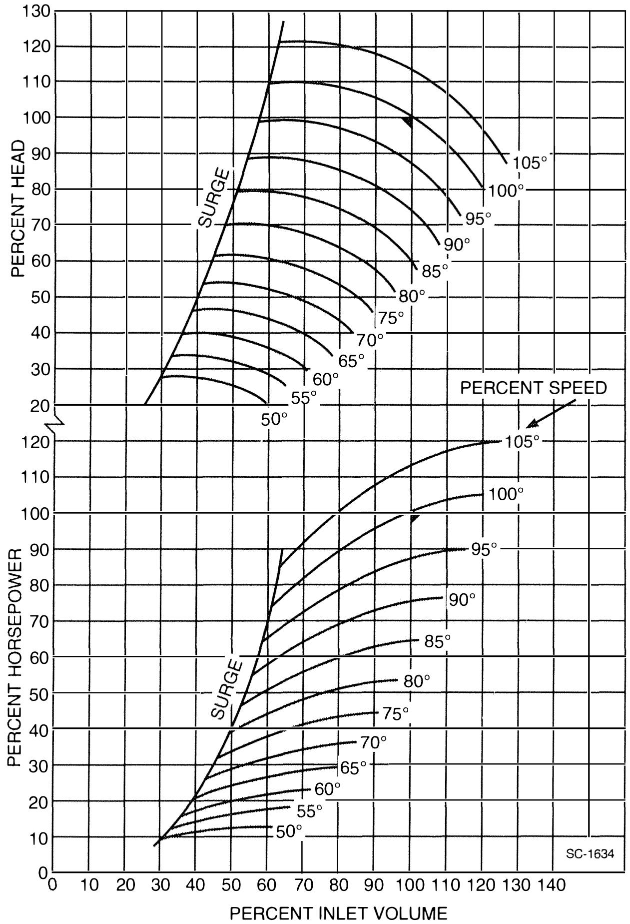


Figure 2. Typical Single Stage Compressor Performance Curve with Speed Control.

The technology exists for the use of conventional design induction or synchronous motors with a control device ahead of the motor to accept 50 or 60 Hz line frequency, converted to DC, and then inverted to an adjustable frequency (and voltage), which is fed to the motor. These controls are expensive, especially in larger sizes above 500 hp. Also, there is an energy loss associated with the drive of two to five percent. In order to justify this type of equipment, a substantial amount of operation at partial speed is required to develop power savings that can justify the initial capital investment. This plant met these criteria, and thus a variable speed induction motor was chosen.

The compressor system consisted of a single stage overhung centrifugal compressor, rated at 21,000 acfm with a pressure rise of 9.5 psig. The compressor operated at a maximum continuous speed of 8,620 rpm. A speed increaser with a ratio of 4.897:1 was used between the motor and the compressor. The gear ratio was chosen, so that full motor speed of 1,785 rpm would operate the compressor at 8,740 rpm, or 102 percent of maximum continuous speed. The gear was rated for 1,030 hp. Both high speed and low speed shafting were coupled by dry disc type couplings. The motor was a squirrel cage induction motor rated at 1,030 hp, which corresponded to the maximum torque capacity of the variable frequency controller.

The current source variable frequency drive that was chosen was one of proven design. The system consisted of two parallel six pulse rectifier inverter circuits that were coupled and phase shifted to produce a 12 pulse AC waveform to the motor. An electrical system schematic diagram is shown in Figure 3.

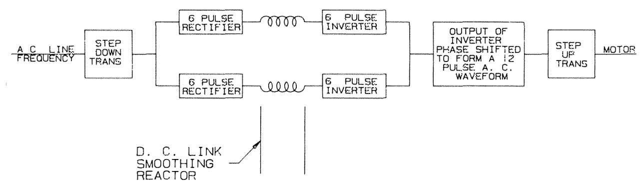


Figure 3. Variable Speed Controller Block Diagram.

The compressor, gear, and motor were all equipped with proximity vibration probes and monitors. All bearings in the train were equipped with resistance temperature detectors and monitors. The compressor, gear, and motor were mechanically tested independently at their point of manufacture.

INSTALLATION, STARTUP, AND PROBLEM SYMPTOMS

The equipment was installed in midyear 1989. During the installation, customer service personnel from the compressor and motor/drive manufacturers were at the site and observed normal installation inspections. The drive was "tuned" to the best ability of the service personnel under no load and partial load operation. The startup phase, with service personnel present, did not allow full speed/full load operation.

The equipment operated satisfactorily through the late summer and early fall of 1989. In early November 1989, plant operation was extended, which required increased operating speeds of the compressor. At approximately 88 to 90 percent of rated speed, an increase in vibration was experienced on the low speed shaft of the gear. At speeds up to 85 percent of rated speed, the low speed gear shaft vibration was less than 0.5 mil peak-to-peak. As the speed approached 88 percent of full speed, the gear vibration would increase beyond the 2.5 mil alarm set point, and as speed was increased between 88 to 90 percent of full speed, the low speed gear shaft vibration would increase to 4.0 mils and trip the driver. None of the other vibration probe levels were experiencing this abnormal increase in vibration.

The problem could be duplicated with clocklike regularity when the unit was restarted and increased in speed. Personnel from a vibration monitoring systems manufacturer were requested to obtain data from the low speed gear probe at as many speeds as possible. Limited data were obtained at two motor speeds, 1,548 rpm (86.7 percent full speed) and 1,572 rpm (88.1 percent full speed). The plant was onstream, and only limited speed increases were permitted to prevent a trip which would result in a process upset. The gear shaft probe spectrum at 1,548 rpm (Figure 4) and 1,572 rpm (Figure 5) contained expected data at 1.0/rev and 2.0/rev; however, there were significant magnitudes at frequencies which were supersynchronous to the gear speed.

SPECTRUM ANALYSIS

The spectrum, taken at a motor speed of 1,548 rpm, had two frequencies that could not be readily correlated with normal mechanical type problems. The first supersynchronous frequency occurred at 2,052 cpm with a vibration level of 0.09 mils peak-to-peak. The second supersynchronous frequency occurred at 2,892 cpm with a displacement of 0.11 mils peak-to-peak.

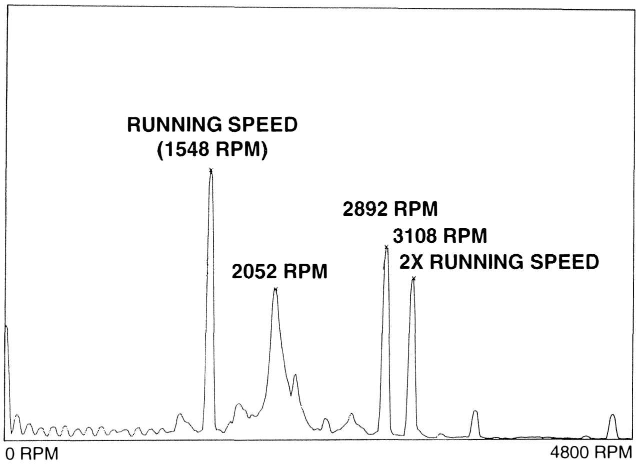


Figure 4. Bull Gear Vibration Spectrum at 1,548 RPM.

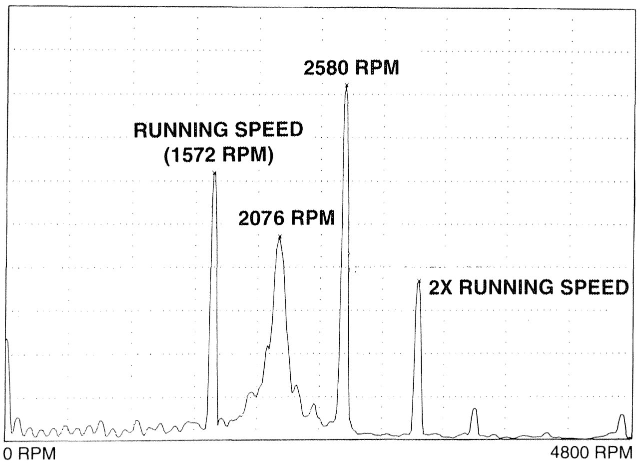


Figure 5. Bull Gear Vibration Spectrum at 1,572 RPM.

The spectrum, taken at a motor speed of 1,572 rpm, revealed that the first supersynchronous frequency had increased to 2,076 cpm, and the displacement had risen to 0.12 mils peak-to-peak. The second supersynchronous frequency reduced to 2,580 cpm, with an increase in displacement to 0.2 mils peak-to-peak.

While the difference in the speeds where these data were taken was only 24 rpm, a change in speed of only 1.5 percent, the two different speeds provided very valuable data, although at the time it could not be readily understood. The most interesting component of the data was the frequency of the second supersynchronous frequency. The fact that the frequency reduced as speed was increased was most unusual and almost never experienced with this type of machinery. While the increment in the frequency for three of the components was the same as the change in motor speed, the second supersynchronous frequency had changed by 312 cpm, or 13 times the speed change of 24 rpm. After some deliberation, only one mechanism was recalled that exhibited a reduction in excitation frequency as the machinery speed was increased. This was the torsional oscillations associated with asynchronous operation of synchronous motors [1]. The excitation frequency created by a synchronous motor is

$$f_e = 2\, f_L \left(\frac{N_{sync} - N}{N_{sync}}\right) \tag{1}$$

where f_e = excitation frequency (Hz)
f_L = system electrical line frequency (Hz)
N_{sync} = synchronous motor speed (RPM)
N = motor speed at any instant in time during acceleration (RPM)

If one tries to use Equation (1), there is no correlation with the data; however, the reduction of frequency with the increase in speed is a consistent concept. The test data were manipulated in various ways until the following equation had evolved.

$$f_{vibration} = (X)\,(60)\,\left[f_L - \left(\frac{N}{N_{max}} \times f_L\right)\right] \tag{2}$$

$f_{vibration}$ = frequency found in the spectrum (cpm)
X = constant
f_L = electrical line frequency (Hz)
N = motor speed at any time (RPM)
N_{max} = maximum motor speed (RPM)

When this equation was applied to the test data, a very interesting observation was made when the second supersynchronous frequency was evaluated

$$2892 = X\,(60)\,\left[60 - \left(\frac{1548}{1785} \times 60\right)\right]$$

X = 2892/477.98 = 6.05

The value of X was very interesting. The drive used for this system was a 12 pulse current wave form system. One would expect to see a factor of 12 in the data; however, the integer six could have some significance as both six pulse and 12 pulse are present in the controller.

When the second data point was evaluated at 1,572 rpm, motor speed

$$2580 = X\,(60)\,\left[60 - \left(\frac{1572}{1785} \times 60\right)\right]$$

X = 2580/429.6 = 6.005

The correlation to the second data point was obviously more than coincidence. It was also realized that the quantity

$$\left(\frac{N}{N_{max}} \times f_L\right)$$

was the same as the output frequency converter; therefore,

$$f_{vibration} = (6)\;(60)\;(f_L - f_{DO}) \tag{3}$$

where

f_L = electrical line frequency (Hz)
f_{DO} = output drive frequency (Hz)

After this discovery was made, further review of the compressor system revealed that the system had a first torsional natural frequency of 1,814 cpm, which was close to the magnitude of the first supersynchronous frequency of the test data.

HYPOTHESIS FOR THE SOURCE OF THE PROBLEM

A hypothesis for the gear vibration was developed. It appeared that the motor was developing an oscillating torque whose frequency was defined by Equation (3). When this frequency coincided with the system's first torsional natural frequency, a condition of resonance occurred, which coupled laterally with the low speed gear shaft. Szenasi [2] documents the previous occurrences of torsional-lateral coupling. Previous work [1] shows a compressor system similar to this has a magnification factor in torsional resonance of 8.0 to 12.0. There is no analytical method to evaluate the coupling of a torsional vibration to a lateral vibration.

The motor/drive manufacturer was informed of the hypothesis that had been developed and was given the test data. The manufacturer agreed that the fact the integer 6, which fit Equation (3), was very interesting, when combined with their knowledge of the drive design.

Service personnel from the drive manufacturer visited the job site to inspect the controls and retune the system. They were given limited access and flexibility in adjusting the controls with the compressor coupled because of plant limitations. The motor was uncoupled and operated to full speed without problems. Likewise, the motor and gear were operated to full speed without any abnormal gear vibrations. Some minor adjustments were made to the drive control; however, the problem persisted.

At this point, there were two paths to consider. One was to instrument the equipment and obtain data to gather further insight into the problem. The other was to estimate the magnitude of the problem and purchase an elastomeric coupling which would add system damping, hopefully to a degree that would eliminate the gear vibrations. When the costs of both alternatives were obtained, the coupling cost was 1/10 of the estimated test cost, and there would be further expenditures after the test to resolve the problem.

A specification for the coupling was developed. The magnitude of the oscillating torque, which was required to properly engineer the coupling, was unfortunately unknown.

The motor/drive manufacturer was requested to quantify the magnitude of the torque oscillations. They estimated that the torque would probably be one to two percent of rated torque; however, they had no analytical or test data to confirm this. As the system magnification was expected to be approximately 10, the specification for the oscillating torque was set to be ± 10 percent of rated torque at a frequency that could be from zero to 720 Hz [3, 4].

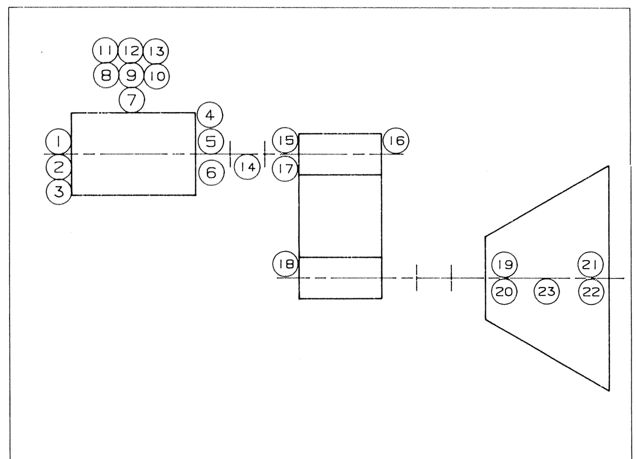
THE FIRST FIELD MODIFICATION

The coupling was procured and installed in June 1990. When the system was restarted, it appeared to operate as before until the motor speed approached 85 to 86 percent speed, at which time the high vibration from the gear low speed shaft probe tripped the unit.

When the unit was restarted, a new problem developed. The unit would operate normally up to 65 percent speed when high gear vibration shut the unit down. Further observation near 60-65 percent speed revealed that the motor and low speed gear shaft were shuttling axially with the motor shaft moving approximately 1/8 in. A compressor service representative was dispatched to the job site. Inspection of the coupling and how it was installed revealed no obvious reason for the new vibration at 65 percent speed. This new problem would place an unacceptable limitation on plant operation; therefore, the elastomeric coupling was replaced by the original dry disc coupling. The system was restarted, and operated acceptably as before to 85 percent speed. The vibration would increase on the low speed gear shaft as 85 percent speed was exceeded.

Obviously, the cause of the problem, or perhaps the magnitude of the excitation, was not understood. A decision was made to instrument the system to obtain sufficient data to resolve the problem. A meeting was held with participation from the user and his electrical consultant, the process contractor, the motor and drive manufacturer, and compressor design personnel. The meeting discussion raised a number of items of concern, from which was developed a test plan.

The author's company, together with an engineering and testing service company, developed an overall test plan to accumulate data that would eventually lead to a resolution of the problem. The compressor train would be instrumented to monitor motor terminal instantaneous current and voltage and the resulting wattage from which motor airgap torque could be evaluated. Thirteen displacement probes and two accelerometers would be monitored, in addition to the low speed coupling spacer strain gauge. All the data points would be recorded on a 24 channel tape recorder. Selected data points could be displayed on an oscilloscope, real time analyzer, or be plotted on an eight pen recorder. Hard copy could be obtained of spectrum plots. The sketch of the test instrumentation is shown in Figure 6, and a more detailed sketch of measured motor terminal parameters is shown in Figure 7.



ITEM NO.	DESCRIPTION
1	MOTOR AXIAL PROBE
2,3	MOTOR OUTBOARD X,Y PROBES (B/N)
4,5	MOTOR INBOARD X,Y PROBES (B/N)
6	MOTOR INBOARD ACCELEROMETER
7	WATT TRANSDUCER
8,9,10	MOTOR CURRENT
11,12,13	MOTOR VOLTAGE
14	COUPLING STRAIN GAUGE
15	GEAR VERTICAL PROBE (B/N)
16	GEAR AXIAL PROBE (B/N)
17	GEAR ACCELEROMETER
18	PINION VERTICAL PROBE
19,20	COMPRESSOR X,Y PROBES COUPLING END (B/N)
21,22	COMPRESSOR X,Y PROBES IMPELLER END (B/N)
23	COMPRESSOR AXIAL PROBE (B/N)

Figure 6. Test Data Point Schematic.

THE FIRST FIELD TEST

The plant was shut down for a turnaround in early August 1990. The compressor piping was altered to allow testing of the compressor on air. The inlet was open to atmosphere, and a discharge throttle valve was installed on the compressor discharge flange to control the compressor load. Engineers were present from the drive manufacturer to attempt further tuning of the drive controls to minimize the vibration. After two days of testing and control board modification, it was finally possible to operate the compressor through its full speed range to 100 percent speed. The gear vibration was continuously monitored and was observed to peak at 90.6 percent rated speed, with a maximum gear vibration of 2.0 to

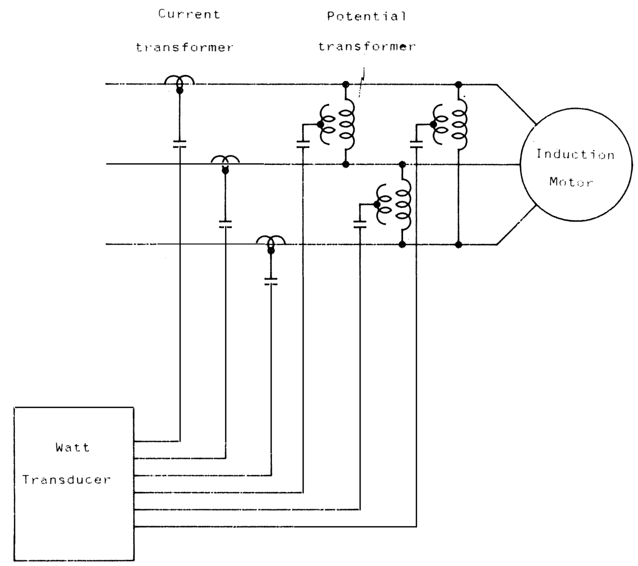


Figure 7. Test Watt Schematic for Measuring Motor Power.

2.5 mils. This was the first time the compressor was operated at full speed, and the first time no unusual noise was heard from the gear at high speeds.

Previously, when the speed approached the point at which the motor would trip due to high gear vibration, a loud "clatter" was heard emanating from the gear.

The drive contained a speed gap feature which was activated. It was found that when the speed gap points were set to 90.6 percent speed ±0.5 percent, the motor accelerated through the point of resonance such that the maximum gear vibration was approximately 1.5 mils, and the gear operated without a vibration alarm or audible noise. All were optimistic that the problem had been resolved. Seeing how plans were made to instrument the system, the test plans continued to document the operating parameters. The coupling spacer was removed, strain gauged, and calibrated. Two days later, all the instrumentation was in place, and a test program defined.

The test program established three points where there had not been any gear vibration problems. The unit was set at 60, 70, and 80 percent speed until sufficient data had been accumulated. After these data were acquired, additional data were collected at much smaller speed increments. Data were collected at 83.3, 86.7, 90.2, and 90.6 percent speed. The maximum oscillating torque read from the strain gauge occurred at 90.625 percent speed. The drive engineer gave some consideration to Equation (2) and, seeing how the wave form to the motor was a 12 pulse current wave form, he was curious if Equation (2) evaluated at X = 12 would produce the large oscillations as well at a different speed. When this was done using the same vibrational frequency as was found in this test

$$F_{vib} = (6)\ (60)\ \left[60 - \left(\frac{90.625}{100}\ 60\right)\right] = 2025 \text{ cpm}$$

At 12 pulses per cycle

$$f_{vib} = 2025 = (12)\ (60)\ \left[60 - \frac{\%N}{100}\ (60)\right]$$

% N = 95.3

To see if this might occur, three additional data points were established at 91.1, 95.3, and 100 percent speed. When the data were collected, it was found that an alternating torque did occur at 95.3 percent speed. In fact, the measured torque was higher than that found at 90.6 percent speed. A tabulation of the test points and the mean and alternating torques measured by the strain gauge are found in Table 1.

Table 1.

TEST SUMMARY

Run #	Drive Output Frequency	% Motor Speed	Drive Set Point	Mean Torque	Peak to Peak Oscillating Torque	Frequency of Peak Amplitude
Test 1						
	Hz		volts	in. lb.	in. lb.	cpm
11	36.0	60.0	4.80	10661.4	7107.6	2100
12	42.0	70.0	5.60	14215.2	21322.8	2070
13	48.0	80.0	6.40	19545.9	3553.8	1980
14	50.0	83.33	6.67	21322.8	5330.7	1950
15	52.0	86.67	6.93	23099.7	3553.8	1980
16	54.1	90.17	7.21	26653.5	15992.1	2010
17	54.375	90.625	7.25	24876.6	42645.6	1860
18	54.65	91.08	7.29	26653.5	21322.8	1770
19	57.187	95.31	7.63	28430.4	51530.1	1710
20	60.0	100.0	8.00	31984.2	2843.	1710
Test 2						
21 & 22	Constant ramp rate acceleration					
Test 3						
23	50.0	83.33	6.67	23099.7	3553.8	
					10661.4	
24	52.0	86.67	6.93	23988.1	3553.8	
					12438.3	
25	54.1	90.17	7.21	26653.5	44422.5	
Test 4						
26	54.1	90.17	7.21	24876.6	17769.	
27	54.375	90.625	7.25	24876.6	46199.4	
					10661.4	
28	54.65	91.08	7.29	24876.6	17769.	

The rated torque of the motor and drive control is

$$T = \frac{63025 \text{ Hp}}{N} \tag{4}$$

T = torque (# in.)
Hp = horsepower
N = motor speed (rpm)

Therefore the rated torque is

$$T = \frac{63025\,(1030)}{1785} = 36367.4 \text{ in. \#}$$

The peak-to-peak alternating torque found at 90.6 percent speed was 42,645 in-lb (117 percent rated torque), and the torque found at 95.3 percent speed was 51,530 in-lb (142 percent rated torque). Obviously, the problem had not been resolved. Seeing how the steady state test points found high oscillating torques at 70, 90.6, and 95.3 percent speed, it was desirable to see if even more locations of excess oscillating torque might be found.

The drive had the capability of a controlled acceleration ramp rate from zero to 100 percent speed. The ramp control was set for zero to full speed in two minutes with a linear rate of acceleration (Figure 8). Numerous locations of high oscillating torque were found, the worst being at startup and those at 90.6 and 95.3 percent speed. The test data revealed the reason for a very loud clattering of the gear teeth until about 10 percent speed was achieved.

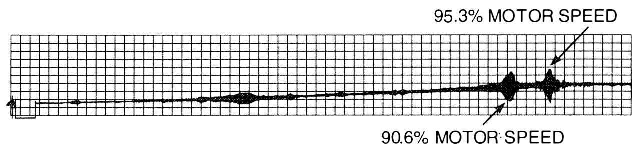


Figure 8. First Field Test Speed Torque Data from Low Speed Spacer Strain Gauge.

These data had revealed that there were high oscillating torques in the system which occurred when the difference frequency multiplied by either a factor of six or 12 coincided with a frequency of 2,025 cpm. The design analysis had calculated a first mode torsional natural frequency of 1,814 cpm. Spectrum data of the strain gauge at the first torsional resonant point, 90.625 percent of rated motor speed, had the maximum amplitude at a frequency of 31 Hz (1,860 cpm), as shown in Figure 9. At the second resonant point of 95.31 percent of rated motor speed, the maximum strain gauge spectrum amplitude occurred at 28.5 Hz (1,710 cpm). These frequencies were within good agreement with the calculated frequency. There was no explanation for the frequency difference found at the two motor speeds.

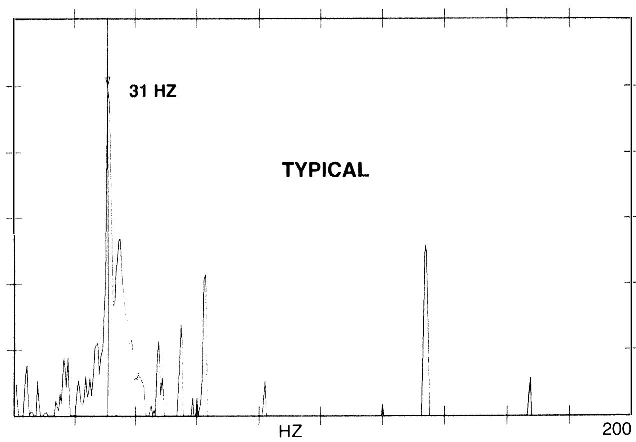


Figure 9. Low Speed Strain Gauge Spectrum at 1,621 RPM.

The modal configuration of the system, Figure 10, revealed a maximum relative amplitude at the motor, indicating that if a oscillating torque did exist at the motor, the system could be driven to a condition of resonance. With little damping in the system, the unit could have a magnification factor of approximately 10 [1]. It then became apparent the frequency that could not be accounted for on the vibration data (Figures 3 and 4) was the first torsional resonance of the system. It is interesting to note that the frequency at which the peak spectrum amplitude occurred did not remain constant, but rather decreased as motor speed increased. A summary of these values is found in Table 1.

The test had been highly successful in both qualifying and quantifying the problem. Before it was terminated another topic was addressed—how successful the tuning efforts had been in lowering the torsional oscillations. To resolve this, a test was conducted after the control boards were retuned to their original configuration, the original startup "tuned" settings. Three test

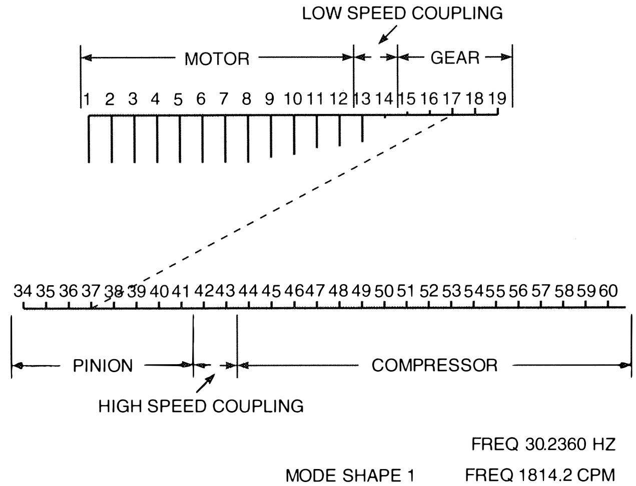


Figure 10. First Torsional Natural Frequency Mode Shape Plot.

points were identified that would slowly approach the 90.6 percent speed where the vibration was the largest. Data were taken at 83.3, 86.7, and 90.2 percent speed. The torsional oscillations at 90.2 percent speed were nearly 275 percent of those with the optimized tuned circuit boards. No further data were gathered. If the 275 percent increase also applied to the resonant speed, the oscillating torque at the first resonance point of 90.6 percent speed would have been nearly 120,000 in-lb peak-to-peak, nearly 325 percent of rated torque. The oscillations about the mean torque would result in a positive torque of 96,370 in-lb, and a negative torque of minus 23,630 in-lb. These data are summarized in Table 1. It is known that a negative torque existed because of the reports of gear box clatter when the unit tripped.

The control boards were reset to the tuned configuration, and test points again taken at 90.2, 90.6, and 91 percent speed to verify the settings had reduced the torque. These tests agreed with the original improvement.

A meeting was held with the user. In order to prevent excessive torsional stresses,the drive would have to be set with speed gaps at 90.5 percent speed ± 1.0 percent, and 95.5 percent ± 1.0 percent. This almost blocked out all operation above 90 percent speed, which was not acceptable to the user. The drive manufacturer was instructed to modify the control with the aid of the test data.

At this point, a very important observation has to be made. If the test data were not taken, would the "tuned" system have been accepted as fixed when the gear vibration was lowered to a peak of 2.5 mils, and the gear clatter eliminated? Had this been the case, and the unit operated at or very close to resonance over a period of time, there surely would have been a fatigue failure in one or more power train components.

One of the objectives of the test was to determine an external measurement that would be an economical, yet reliable indication of torsional excitation. Previous work that had been done on synchronous motors determined that a Hall effect watt transducer would be a reliable indicator of the frequency and magnitude of the difference between the direct and quadrature axis torques [1]. In some instances, this measurement has been a requirement of synchronous motor acceptance tests.

TEST DATA ANALYSIS AND SYSTEM ANALYSIS

A wealth of data was recorded and evaluated at all test points, but is presented here for brevity at the first resonance of 90.6 percent speed (1,621 rpm). A signature of the motor terminal voltage is shown in Figure 11, and a time domain plot of the motor

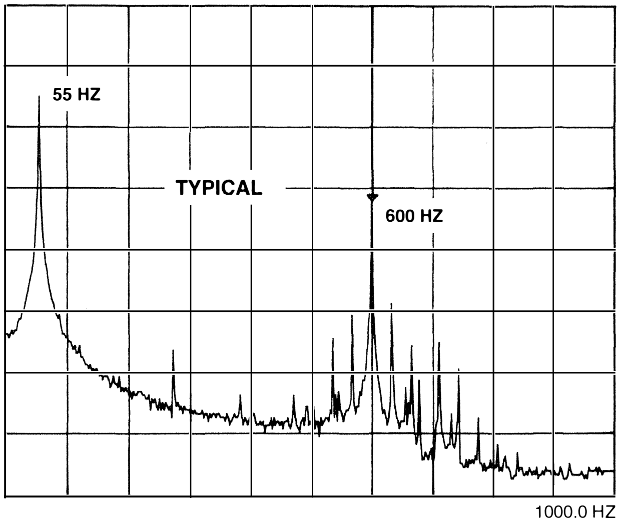


Figure 11. Motor Terminal Voltage Spectrum at 1,621 RPM.

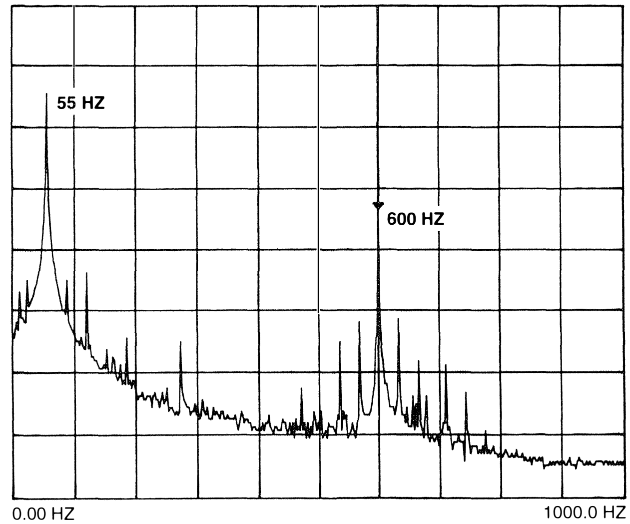


Figure 13. Motor Terminal Current Spectrum at 1,621 RPM.

voltage is shown in Figure 12. The characteristics of the 12 pulses/cycle are evident. A spectrum plot of the motor current is presented in Figure 13. The peak occurs at the same frequency as that found in the voltage spectrum. A time domain presentation of the motor current with a characteristic 12 pulse per cycle is shown in Figure 14. A spectrum plot of motor kilowatts is presented in Figure 15. No obvious indication of the torsional resonance is apparent in these data. The spectrum of the strain gauge signal, Figure 9, shows a very predominant frequency at 31 Hz, the torsional natural frequency of the system. This approximate frequency was found in all the strain gauge test point spectrums. There was no other sensor that was an accurate indicator of the torsional vibration in the compressor system.

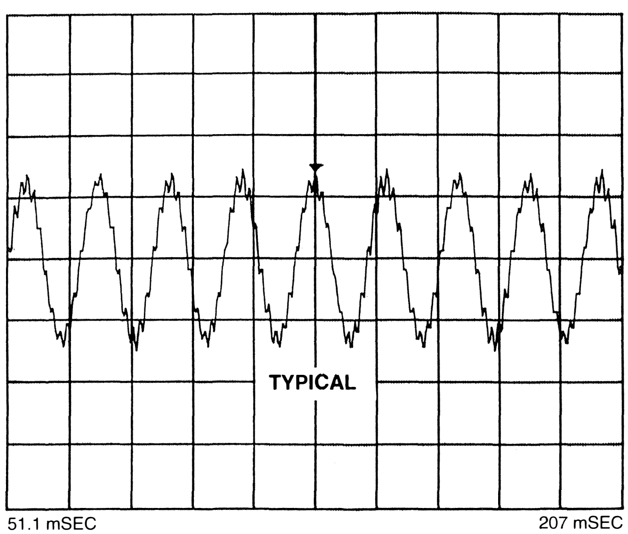


Figure 12. Motor Terminal Voltage Time Domain at 1,621 RPM.

The test data were reduced to a report of 92 pages which documented the various sensor data in time domain, spectrum, and cascade plots. The data were forwarded to the drive vendor for analysis. In addition, a simplified torsional model of nine inertias and seven torsional spring constants was forwarded (Figure 16). This simplified model was condensed from an ANSYS finite element system model of 50 nodes which consisted of dimensional

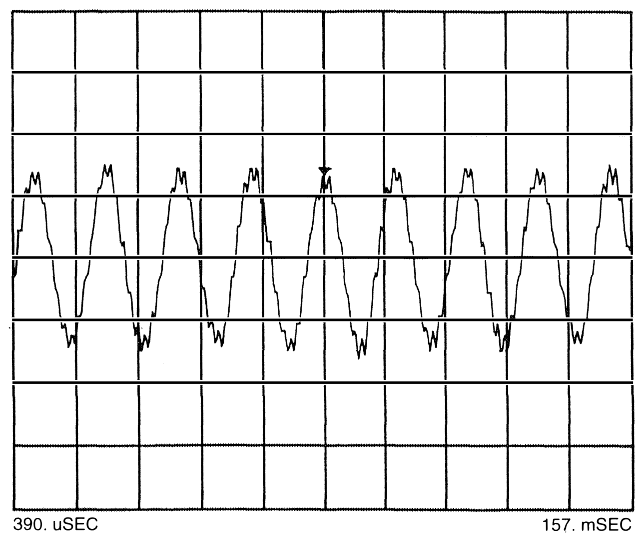


Figure 14. Motor Terminal Current Time Domain at 1,621 RPM.

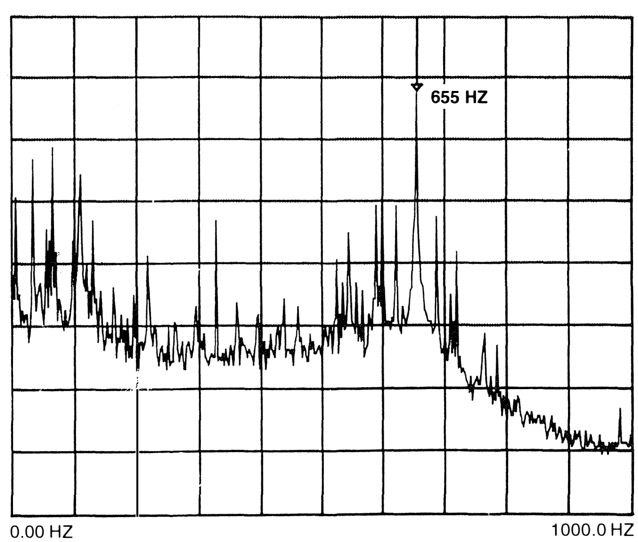
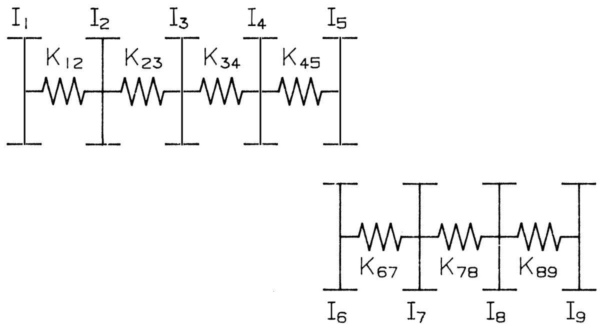


Figure 15. Watt Transducer Spectrum at 1,621 RPM.

shaft elements, lumped inertias, and lumped spring constants. The summarized model was reanalyzed with ANSYS and found to have the first two mode frequencies within three-fourths of one percent of the full model. In addition, the mode shape and controlling kinetic and strain energies occurred in similar locations in both models.



I_1	2147.8 * IN^2		K_{12}	12.18×10^6 IN LB/RAD
I_2	62585.7 * IN^2		K_{23}	13.48×10^6 IN LB/RAD
I_3	770.9 * IN^2		K_{34}	11.50×10^6 IN LB/RAD
I_4	791.3 * IN^2		K_{45}	34.64×10^6 IN LB/RAD
I_5	18782.9 * IN^2		K_{56}	6.20×10^6 IN LB/RAD
I_6	198.1 * IN^2		K_{67}	2.87×10^6 IN LB/RAD
I_7	132.7 * IN^2		K_{78}	3.13×10^6 IN LB/RAD
I_8	132.7 * IN^2			
I_9	7641.5 * IN^2			

Figure 16. Torsional Schematic of Motor/Gear/Compressor System.

This analytical simulation using this type of simplified model has been used successfully for over 15 years for transient startup simulation analysis of synchronous motor driven compressor systems and has been verified by field testing.

The drive manufacturer created a new electromechanical system simulation model to analyze the system. The model was quite complex; however, it was not detailed enough to accurately simulate the intricacy of the entire electrical drive system and the mechanical system and their interaction. As discussed previously, the spectrum or time domain data on the motor air gap kilowatts did not give a clear indication of the actual "difference" frequency energy. Nonetheless, this new model did confirm a "difference" frequency effect, as expected from Equation (3). A substantial number of simulations were made to evaluate the effect of the electrical drive components on the difference frequency excitation torques. The conclusion of this work was a decision to modify the choke or DC link between the rectifier and inverter segments of the drive.

Two new chokes were manufactured and made available for installation at the next shutdown. In June 1991, the chokes were installed. In order to expedite the test turnaround time, a new coupling spacer was purchased, strain gauged, calibrated, and made available to be installed as required.

SECOND FIELD TEST

The first test performed was an overview to see what changed. A two minute ramp run was conducted and taped. A pen chart of this run is provided in Figure 17. There is considerable oscillating torque at startup; however, it was reduced. The audible "clatter" of the gear box upon startup was not quite as bad as it had been previously. The resonance at 90.6 and 95.3 percent speed were still present. The first test points of the August 1990 test were duplicated by the adjustment of the drive set point voltage. A listing of the data points is presented in Table 2. It can be seen that the resonant

Table 2.

1st Test Number	2nd Test Number	Drive Out-Put Freq. Hz	% Motor Speed	Drive Set Point Volts	Mean Torque In #	Oscillating Torque In # Peak to Peak	Motor Current Amps	Gear Vibration Mils	Compressor Vibration Mils, X/Y
11	1A	36	60	4.809	6092	—	165	0.5	2.8/2.3
12	2A	43.5	72.5	5.8	10661	—	213	0.5	2.7/3.4
13	3A	48	80.0	6.404	13707	—	252	0.5	2.5/3.25
14	4A	50	83.33	6.678	15230	—	275	0.5	2.4/3.2
15	5A	52	86.67	6.93	15230	6092	295	0.6	2.5/3.1
16	6A	54.1	90.17	7.21	18276	48738	317	3.0	2.5/3.0
17	7A	54.375	90.625	7.25	18276	18276	319	1.2	2.5/3.0
18	8A	54.65	91.08	7.29	16753	18276	325	1.1	2.4/2.9
19	9A	57.187	95.31	7.63	19789	9138	355	.5/.8	2.1/2.7
20	10A	60.0	100	8.00	21322	9138	395	.6	1.75/2.0

point had shifted slightly. This probably was the result of a slight difference in the as machined characteristics of the new spacer spool. The stiffness of the low speed coupling spring constant was most influential on the first torsional natural frequency. It contains 39.5 percent of the strain energy for the first mode [4]. The mean torques were somewhat lower as well; this was probably due to the setting of the discharge throttle valve. The first peak resonance had shifted from 90.6 percent to 90.2 percent speed; however, the oscillating torque was measured to be 48,738 in-lb peak-to-peak.

The new choke had cleaned up the startup torque trace in many locations; however, it did not accomplish what had been anticipated. The same upper limit restriction was left on the drive at 87.5 percent rated speed.

ENGINEERED COUPLING WITH TORSIONAL DAMPING

The drive manufacturer continued his pursuit of the problem. The drive was of European design; therefore, consultations were made with the European designers. The Europeans acknowledged that they had previously experienced this difference frequency, calling it the ripple frequency. In fact, the problem had been identified [5]; however, the magnitudes were identified as being

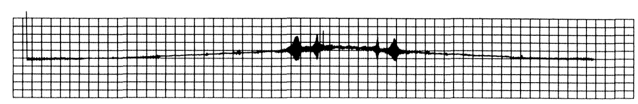


Figure 17. Second Field Test Speed Torque Data from Low Speed Spacer Strain Gauge.

approximately one percent of the motor "quasi-steady state motor torque in air gap." When queried as to how they solved the problem, they responded that they now recommend elastomeric coupling to introduce torsional damping. The drive manufacturer, therefore, recommended the use of an elastomeric coupling. In addition, they intended to pursue ideas on the drive feedback controller in the event the system was feeding upon itself as it entered resonance.

The contractor and user were aghast at the recommendation of an elastomeric coupling, seeing how the first attempt was field tested and failed over a year ago. Alternate methods were demanded to achieve maximum continuous speed. A standard frame size inlet guide vane design was available, and its components were put on order immediately. Also, a request was made that the gear manufacturer size a high ratio gear set that could achieve compressor maximum continuous speed at 85 percent motor speed. This proved to be unfeasible, due to inadequate gear to pinion center distance. The user required that the new coupling be made available for a scheduled August 1991 shutdown, a lead time of seven weeks from the meeting after the June 1991 test.

The original vendor of the elastomeric coupling was supplied with the results of the two field tests and was asked to propose a design and explain why the original coupling was ineffective. Design personnel of the coupling manufacturer replied that the original coupling contained urethane blocks which have little or no damping ability. They had other features such as being nonlubricated and resistant to abrasion; however, they were a complete misapplication for a system requiring torsional damping. They did have a resilient coupling design, but could not meet the fleeting seven week shipment.

Another vendor was contacted, advised of the problem, and supplied with the test data. With bilateral cooperation between the companies, the coupling was shipped within seven weeks of the meeting. The spacer of the coupling was shipped to the engineering and testing service company for strain gauge installation and calibration.

The scheduled plant shutdown and turnaround was delayed until early October 1991. The low speed coupling was removed, and the key seat areas of both the motor and gear were dye penetrant inspected to ensure that no cracks had formed from the limited operation near or at torsional resonance. The new coupling was installed with the spacer instrumented as before.

THIRD FIELD TEST

There were some initial instrumentation problems with the strain gauge signal due to the antenna location. This problem was rectified and the test started. The original startup consisted of test points at 10 percent speed intervals until full speed was reached. The gear was considerably quieter; however, there was audible low speed clatter. None of the test points resulted in gear vibration that exceeded 1.0 mil. No gear vibration alarms occurred from zero to 100 percent speed. Unfortunately, the strain gauge signal revealed an oscillating torque at all test points of approximately 15,000 in-lb peak-to-peak. This was present consistently and was significantly higher than the off resonance oscillating torques recorded with the previous disc type coupling. A ramp test was conducted as before. An oscillograph chart of the startup is shown in Figure 18. The trace shows some low speed resonance and a blossom near full speed. The peak oscillating torques that were recorded were nearly 32,000 in-lb peak-to-peak. The test was terminated with disappointing results.

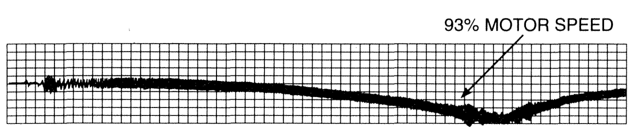


Figure 18. Third Field Test Speed Torque Data from Low Speed Spacer Strain Gauge.

The results were discussed at length to determine an explanation for the test data. The supplying vendor was contacted to see if they had ever experienced similar results; however, they had no explanation for this data. An improper ground on the strain gauge was suspected. With the spacer mounted on rubber blocks, it would be virtually isolated from a common machinery ground. The plant was asked to check the resistances at various locations on the coupling and the adjacent shaft extension (Figure 19). The data confirmed the suspicion, and a shop test was set up to determine the grounding effects.

A section of pipe was strain gauged and set upon a pair of pipe rollers. As the pipe was rotated, there was no extraneous torque

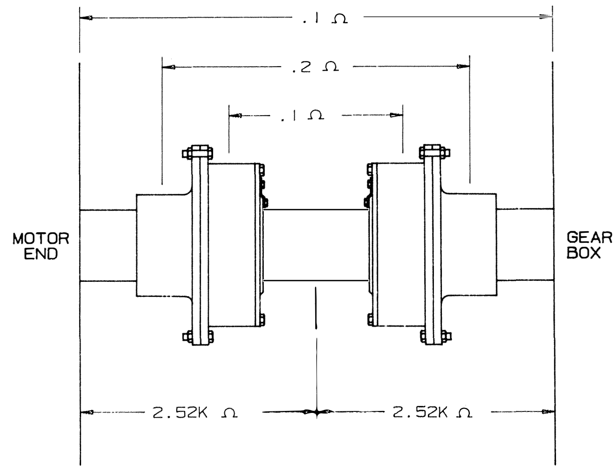


Figure 19. Low Speed Elastomeric Coupling Electrical Resistance Measurements.

indicated by the strain gauge signal. The pipe ends were wrapped with rubber sheeting and taped. When the pipe was set on the rolls and rotated, the signal recorded in the shop was almost identical to the measurements recorded in the field.

A retest was scheduled at the back end of the shutdown. Ground straps were added to "jump" the rubber blocks, as shown in Figure 20. The retest data was very encouraging when the motor was tested at 10 percent intervals to 80 percent speed and then in finer increments (Table 3).

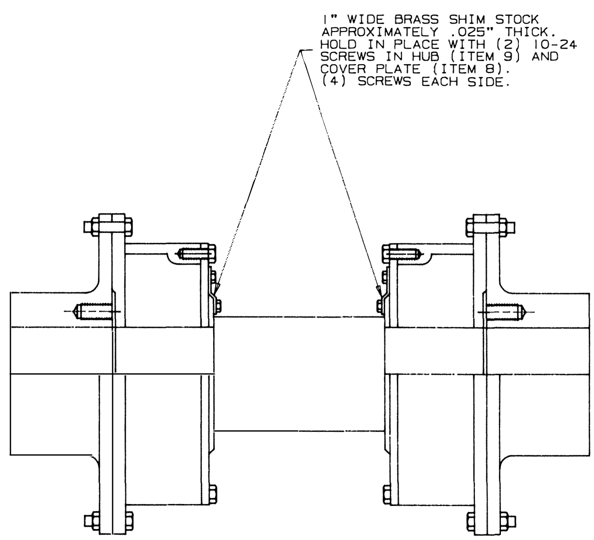


Figure 20. Grounding Straps for Low Speed Elastomeric Coupling.

A speed ramp test was conducted, which revealed some minor resonance at startup and a clean signal to approximately 93 percent speed. The added damping modified the resonance so that only one peak was found at 93 percent speed (Figure 21). With the promising results of this test, adjustments were made to the compressor valving such that the drive was loaded to its maximum capacity. Any speed reduction from this point would be representative of the maximum expected loads at reduced speeds. The motor speed was set to operate at 100 percent speed. The discharge valve was adjusted to load the compressor at the rated output of the drive. The motor speed was reduced to 93 percent speed to operate on

Table 3.

Test Data October 27, 1991

4th Test Number	Drive Set Point Volts	Drive Out-Put Freq. Hz	% Motor Speed	Compressor Discharge Pressure	Mean Torque In #	Oscillating Torque In #	Motor Current Amps	Gear Vibration Mils
1C	.877	6.58	10			~3256		0.3
2C	1.60	12.00	20			~3256		0.7
3C	2.40	18.00	30			~3256	127	0.7
4C	3.20	24.0	40		9117.	~3256	158	0.6
5C	4.0	30.0	50	1.2#	12764.	~3256	202	0.5
6C	4.80	36.0	60	2.0#	18235.	~3256	262	0.5
7C	5.602	42.0	70	2.5#	23705.	~4884	331	0.6
8C	6.406	48.0	80	5.0#	27352.	~4884	362	0.6
9C	6.797	51.0	85	6.0#	29176.	4884	403	0.6
10C	7.208	54.0	90	7.0#	32823.	9768.	445	0.75
11C	7.45	55.9	93.2	8.5#	27352.	16280	386	1.2
12C	7.60	57.0	95	9.0	27352.	9768.	399	0.75
13C	8.00	60.0	100	10.5	30999	8140	450	0.70
14C	8.0		100		34646	4884		
19C	7.203	54.0	90		21882.	6512.	364.	0.6
18C	7.281	54.6	91		22794.	6838.	370	0.7
17C	7.36	55.2	92		23706.	9768.	380	0.8
16C	7.441	55.8	93		24435.	16280.	385	1.20
15C	7.517	55.4	94		25529.	11396.	395	1.0
14C	7.60	57.0	95		26258.	9117.	401	0.75

resonance. Here, data were collected at a mean torque that was representative of part speed operation. The peak-to-peak torque measured was 16,280 in-lb. The new coupling resulted in a substantial reduction from 43,000 and 50,000 in-lb to 16,280 in-lb; however, the peak-to-peak oscillating torque still represented 44 percent of the rated drive torque. The vibration data recorded on the gear reached a peak level of 1.2 mils peak-to-peak. No alarms were recorded during the test. The significant changes that had been accomplished are listed in Table 4.

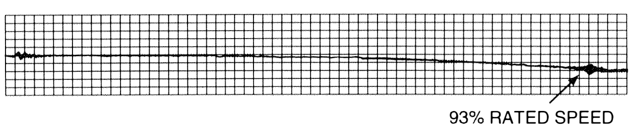


Figure 21. Fourth Field Test Speed Torque Data from Low Speed Elastomeric Coupling.

Table 4.

	1st Test	2nd Test	4th Test
1st Resonance	42646 in. # peak to peak @ 90.6% rated speed	42640 in. # peak to peak @ 90.17% rated speed	16280 in. # peak to peak @ 93% rated speed
	Mean Torque 24877 in. #	Mean Torque 18274 in. #	Mean Torque 24435 in. #
2nd Resonance	51530 in. # peak to peak @ 95.3% rated speed	45685 in. # peak to peak @95.1% rated speed	No definable peak
	Mean Torque 28430 in. #	Mean Torque 21320 in. #	

This appeared to be the best operation that could be anticipated of the compressor system, and no further electrical enhancements could be suggested by the drive manufacturer. While the mechanical performance met the requirements of the user, the long term fatigue life of shafting and gearing had to be proven.

As a starting point, a rigorous requirement of 10 years operation on resonance was investigated. As a preliminary calculation, the gear vendor was given test data and the required 10 years operation. There would be an additional reduction of the torsional oscillation by the gear side of the new coupling; however, the reduction could not be quantified without additional system simulation. The gear vendor evaluated the requirements and compared them to the original design requirements of the gear. The gear teeth were designed for resistance to high cycle fatigue as a norm. With each revolution of the gear, the teeth load rose from zero to the torque requirement of the design and back to zero. The gear was rated for 1,030 hp @ 1,785 rpm or 36,367 in-lb. The test data revealed that the gear torque at resonance would be

$$\text{Gear Torque} = \text{Mean Torque} + \frac{\text{Oscillating Torque}}{2}$$

therefore the torque at resonance would be $24{,}435 + \dfrac{16{,}280}{2}$

or 32,575 in-lb, which is less than the rated torque of the gear. There was an additional oscillating torque at full speed of approximately 5,000 in-lb peak-to-peak, which would be reduced somewhat by the gear side new coupling. The gear vendor anticipated no cause for concern from this additional torque.

SUMMARY

- A radial proximity probe mounted on the low speed gear shaft responded to a torsionally coupled lateral vibration.
- Spectrum analysis of the gear probe vibration signal provided information that resulted in a hypothesis that the lateral vibration was caused by a torsional excitation.
- An initial "fix" was made on an approximated magnitude of the torsional excitation. A nonmetallic coupling material was misapplied to the defined application.
- Field test data identified several speeds at which torsional resonance occurred. The most severe resonances occurred at low speeds during starting and above 90 percent motor speed.
- The torsional resonance was excited by the current source variable frequency drive at a frequency of six and 12 times the difference between the electrical line frequency and the drive output frequency. This excitation was also identified as the beat frequency or interharmonic distortion.
- Field data measurements identified the peak-to-peak torques in the low speed spacer spool at 142 percent of rated drive torque. Measurements indicated that the system, as originally installed, could have had peak-to-peak torques of 325 percent of the rated drive torque.

The presence of negative torques, with the equipment as originally installed, were known to exist as gear "clatter" and could be heard during high vibration trips.

- Modelling of the electronic drive system and the mechanical system identified the existence of the "difference frequency;" however, these calculations could not accurately predict the motor air gap torque.
- The system model did not accurately predict a "fix" for excessive torques developed by the drive.
- It is not possible to design a typical motor-compressor or motor-gear-compressor system with a first mode torsional natural frequency that can avoid the difference frequency.
- The application of a properly engineered elastomeric coupling reduced the resonant low speed coupling spacer torques to approximately one-third of the oscillating torques that occurred with a dry disc coupling.

Two resonant frequencies that occurred with the dry disc coupling were reduced to one torsional resonant frequency, approximately midway between the original torsional natural frequencies, when the new coupling was used.

• A fatigue analysis confirmed that the modified system with the new coupling was suitable for continuous operation anywhere between 20 percent to 100 percent speed.

• The only reliable measurement made during this test that indicated torsional resonance was the strain gauge mounted on the low speed coupling spacer.

CONCLUSION

A source of significant torsional excitation was produced by a current source variable frequency drive which prevented safe and reliable operation of a motor-gear-compressor system. It was determined that, in the case of this drive vendor, accurate analytical methods did not exist to predict the magnitude of the difference frequency torque. Field test data were required to define the actual oscillating torques in the system. Further testing and analysis was required to verify that an engineered solution to the problem would result in equipment that was safe and reliable in a variable speed environment.

It appears that incomplete communication exists between the drive vendor, the motor vendor and the vendor with system responsibility with regard to the frequency and magnitude of torsional oscillations developed at the motor shaft when a variable speed electric drive is used. This information is essential to properly engineer such a system.

Based on this experience, it is apparent that further research and development and subsequent field tests are required to develop a level of confidence in analytically engineered systems. Because this is a system, honest disclosure and mutual cooperation of all parties is required.

The magnitude of this problem was so severe that a problem would have been detected, due to the gear "clatter" while in resonance. Significant torsional vibration may occur without external signs of distress. Once the oscillating torques were reduced so that negative torques were eliminated, the only telltale sign of trouble, other than the coupling strain gauge, was the magnitude of radial vibration found on the low speed gear shaft.

Users operating variable frequency electric motor drives should consider reviewing their system designs. The ultimate failure would manifest itself as high cycle fatigue. In order to evaluate whether a fatigue failure might occur, the operating stress levels (both steady state and dynamic), the frequency of resonance, and the period of time operating in resonance must all be known quantities prior to a fatigue analysis.

REFERENCES

1. Hudson, J. H., "Peak Shaft Torques of Three Mass Systems Operating Near or Passing Through Torsional Resonance," ASME Paper 74-PET-27(1974).
2. Szenasi, F. R. and Wachel, J. C., "Field Verification of Lateral-Torsional Coupling Parts on Rotor Instabilities in Centrifugal Compressors," A Presentation at the 15th Meeting of the Industrially Sponsored Program on the Dynamic Analysis of Turbomachinery, University of Virginia (1980).
3. Sheppard, D. J., "Torsional Vibration Resulting from Adjustable-Frequency AC Drives," IEEE Transaction on Industry Applications, *24*, (5) (September/October 1988).
4. Molnar, A. J. and Wolff, F. H., "Variable Frequency Drives Multiply Torsional Vibration Problems."
5. Simmons, H. R. "Vibration Energy: A Quick Approach to Rotor Dynamic Optimization," ASME Paper.
6. The Journal of the Energy and Automation Group Electrical Installations and Automotive Systems Group and KWU Group of Siemens AG, *X* (October 1988).

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